SUPERCRITICAL RANKINE CYCLE

A synopsis of the cycle, it's background, potential applications and engineering challenges.

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Abstract

The Rankine cycle has been using water to generate useful work since the mid 1800's. With the advent of modern *super alloys*, the Rankine steam cycle has progressed into the supercritical region of the coolant and is generating thermal efficiencies into the mid 40% range. As of 2001, there were 360 supercritical fossil fuel plants operating across the globe [10]. These plants are operating at temperatures around 1000 °F and pressures around 3500 psia. Two of the current engineering challenges are to adapt the supercritical Rankine cycle to coolants other than water, and to design nuclear power plants capable of operating at supercritical temperatures and pressures. There is much work to be done in the area of material development before nuclear plants will be capable of reliable operation at the temperatures and pressures required to achieve thermal efficiencies over 35%.

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The Rankine cycle is synonymous with the steam cycle and is the oldest functional heat cycle utilized by man. A basic diagram of the cycle is illustrated in Figure 1. Figure 2 represents the Temperature - Entropy (T-S) diagram for the simplest version of the Rankine cycle which corresponds to systems with working pressures up to 400 psia [1]. The cycle as it is depicted in Figure 2 is rarely used today.

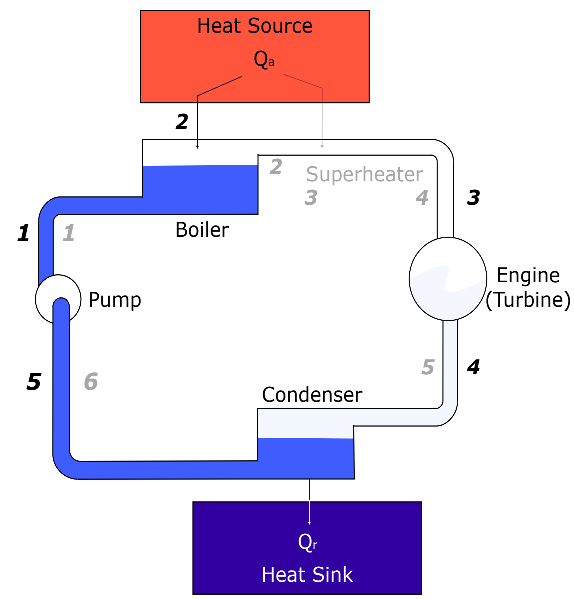


Figure 1. Simple Rankine/Steam Cycle
Black numbers correspond to Figure 2;
Grey numbers correspond to Figure 3.

More commonly, the simple Rankine cycle is modified such that the steam is superheated prior to entering the turbine. This is indicated by line 3-4 in the T-S diagram shown in Figure 3. When properly designed, this modification increases W (the work capacity of the system) without significant increase to Q_r (the unavailable heat of the system). Typically, point 4 represents temperatures up to approximately 900 °F (755 K). Theoretically, it should be possible to superheat the steam to the same temperature as the heat source; which in a boiler, is on the order of 3500 °F (2200 K). However, the temperature that corresponds to point 4 is physically limited to a differential temperature of about 1000 °F (811 K), the metallurgical limit [2]. In other words, so long as the heat transfer across the materials in the boiler and the turbines maintains a temperature differential less than the operational limits of the material, it should not experience catastrophic failure. Additionally, superheating allows the steam to still remain approximately 90% (or greater) dry as it exhausts from the turbine. This feature of the superheated cycle simplifies the turbine design and extends turbine life due to reduced wear from water impingement on the blades.

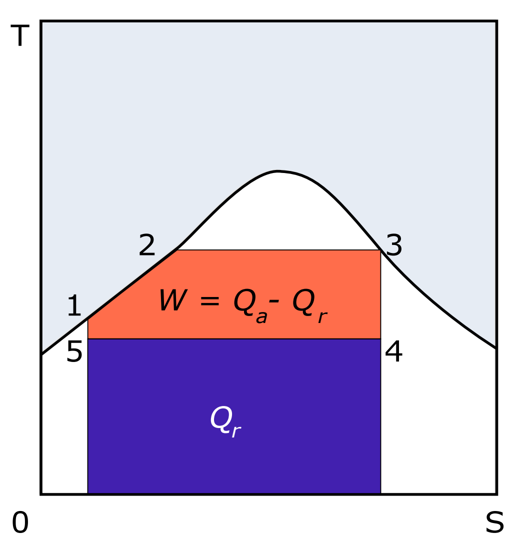


Figure 2. T-S Diagram, Simple Rankine Cycle.

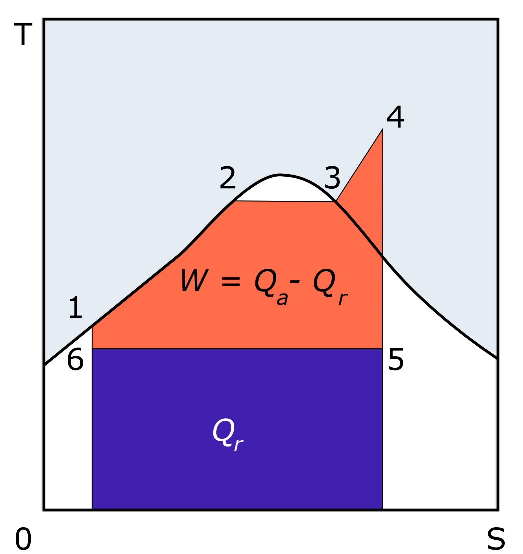


Figure 3. T-S Diagram, Simple Rankine Cycle with Superheat.

The efficiency (η) of this cycle can be represented by the ratio of work capacity divided by the heat put into the system (Q_a):

$$\eta = \frac{W}{Q_a} \qquad \text{Equation 1.}$$

In terms of enthalpy (h), the efficiency for the Rankine cycle with and without superheating can be defined by the following equations respectively:

$$\eta_{\text{simple}} = \frac{h_3 - h_1 - h_4 + h_5}{h_3 - h_1} \qquad \text{Equation 2.}$$

$$\eta_{\text{superheat}} = \frac{h_4 - h_1 - h_5 + h_6}{h_4 - h_1} \qquad \text{Equation 3.}$$

Typical efficiencies for these cycles range from around 20% - 35%.

There are two measurable physical and covariant parameters of the system which correlate to substantial affects on the thermal efficiency of the cycle. The first parameter is the system back pressure. For a given inlet (throttle) temperature and pressure at the turbine, a lower back pressure equates to a higher system efficiency. In effect, an increase in the back pressure reduces the capacity of the steam to expand through the turbine. The effect of variations in system back pressure can be seen in Figure 4. The second parameter is the throttle temperature. While holding both the system back pressure and the throttle pressure constant, an increase in the throttle temperature will increase the thermal efficiency of the cycle. This effect is shown in Figure 5. An increase in the throttle temperature simply increases the potential of the expanding steam to transfer energy to the turbine.

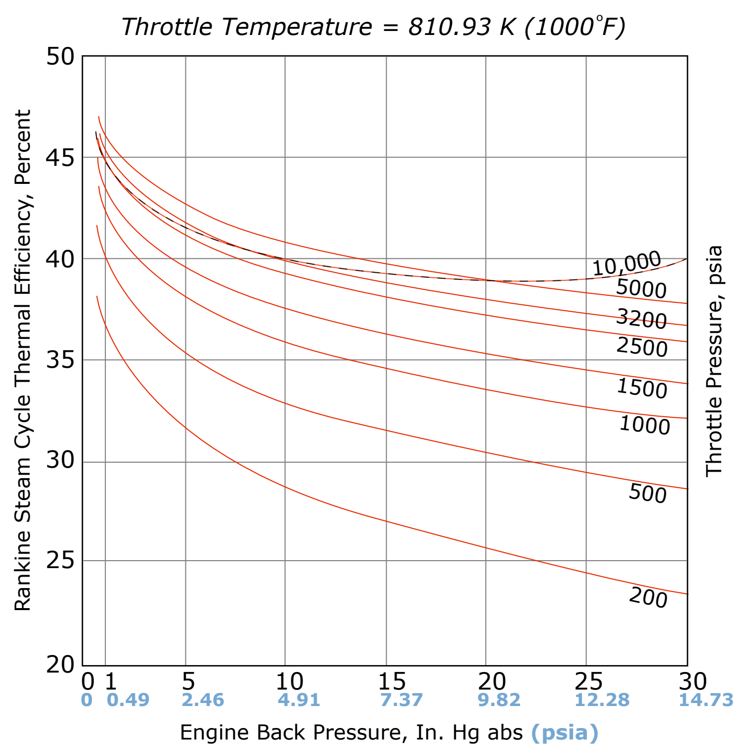


Figure 4. Effect of Turbine Outlet Backpressure for a Constant Input Throttle Temperature [1].

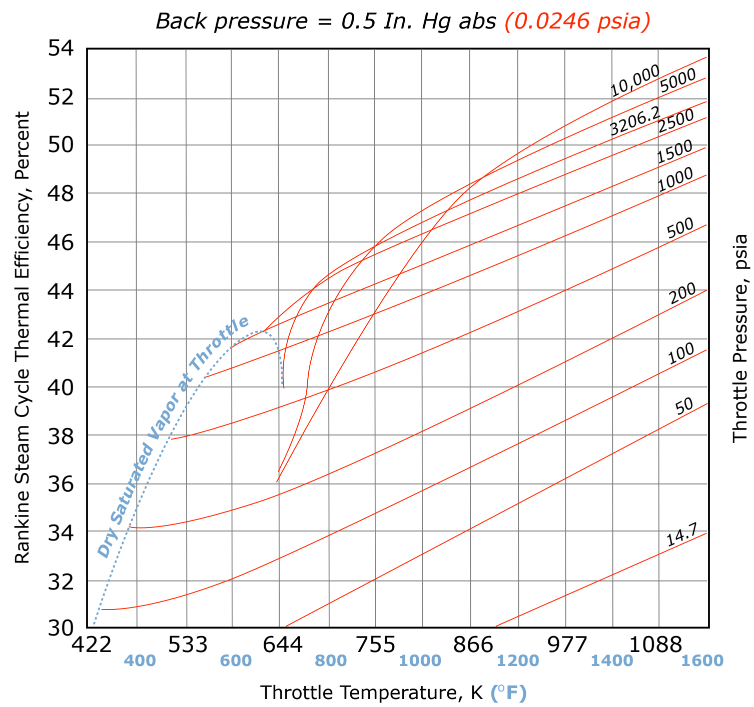


Figure 5. Effect of Turbine Inlet Temperature for a Constant Outlet Backpressure [1].

Supercritical Rankine Cycle

The Rankine cycle can be greatly improved by operating in the supercritical region of the coolant. Most modern fossil fuel plants employ the supercritical Rankine Steam Cycle which pushes the thermal efficiency of the plant (see equation 4) into the low to mid 40% range.

$$\eta_{\text{supercritical}} = \frac{h_2 - h_1 - h_3 + h_4}{h_2 - h_1} \qquad \text{Equation 4.}$$

For water, this cycle corresponds to pressures above 3,206.2 psia and temperatures above 705.4 °F (646.3 K) [1]. The T-S diagram for a supercritical cycle can be seen in Figure 6. With the use of reheat and regeneration techniques, point 3 in Figure 6, which corresponds to the T-S vapor state of the coolant after it has expanded through a turbine, can be pushed to the right such that the coolant remains in the gas phase†. This simplifies the system by eliminating the need for steam separators, dryers, and turbines specially designed for low quality steam.

† *Traditionally, a Rankine cycle operates across a phase change and a Brayton cycle operates entirely in the gas phase. However, as improvements in material make higher temperatures and pressures available, the difference between a supercritical Rankine cycle and a Brayton cycle blur. For the purpose of this paper, a supercritical Rankine cycle is one which has evolved from a simple Rankine cycle or one for which it is desirable to run along the gas/vapor boundary (dimer and trimer formation only) to maximize heat extraction from the coolant.*

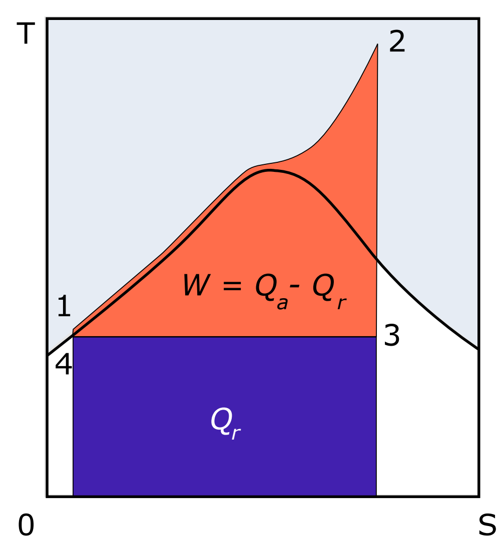


Figure 6. Supercritical Rankine Cycle.

The primary concern with this cycle, at least for water, is the material limits of the primary and support equipment. The materials in a boiler can be exposed to temperatures above their limit, within reason, so long as the rate of heat transfer to the coolant is sufficient to "cool" the material below its given limit. The same holds true for the turbine materials. With the advent of modern materials, i.e. super alloys and ceramics, not only are the physical limits of the materials being pushed to extremes, but the systems are functioning much closer to their limits. The current super alloys and coatings are allowing turbine inlet temperatures of up to 1290 °F (973 K) [3] and fourth generation super alloys with Ruthenium mono-crystal structures promise turbine inlet temperatures up to 2010 °F (1370 K) [4] in the future.

Nuclear Light Water Power Application

The Generation IV SuperCritical Water-cooled Reactor (SCWR) is a 1600 MW_e nuclear power plant design based on the supercritical Rankine cycle. The system layout is identical to the supercritical system currently used in fossil fuels plants except the boiler is replaced with a nuclear reactor. The system diagram can be seen in Figure 7, page 5. Conventional nuclear plants operate on the simple Rankine cycle with superheating and exhibit thermal efficiencies on the order of 33%. The SCWR is capable of a thermal efficiency of 44.8% [5]. One of the attractive aspects of a supercritical cycle for a nuclear application is that it eliminates the need for pressurizers, primary to secondary heat exchangers, steam dryers, and steam generators. When one considers that the primary cost of electricity for a nuclear plant is the construction cost, elimination of support equipment is crucial, especially if it offsets the cost of heavier pressure vessels and support piping.

The primary differences between the SCWR and conventional Boiling Water Reactors (BWR) & Pressurized Water Reactors (PWR) is the reactor outlet temperature is 930 °F (770 K) instead of 570 °F (570 K) and the SCWR operates at approximately 3625 psia [5] as compared to 915 psia and 2265 psia respectively [6]. This difference is graphically represented in Figure 8, page 5. In order to handle the higher pressures and temperatures, the reactor vessel must be designed to handle the extreme temperatures and pressures. The specifications required to meet this demand are listed in Table 1. More important than the pressure vessel however, are the fuel assemblies and control rod components. With a reactor inlet temperature of 535 °F (550 K), the reactor components are exposed to a temperature differential of 395 °F (220 K). Unlike the tubes in a boiler, a significant portion of the fuel assembly structure is not directly transferring heat to the fluid. This places much greater stress on those components and consequently, Phillip E. MacDonald performed a first order stress test on many of the common materials used for these assemblies. Of the austenetic (304L & 316L) and Nickle (Alloy 600, 625, 690,

Table 1: SCWR Pressure Vessel Specifications [5]

Parameter	Value
Material	SA-533 or SA-508 Grade 3, Class 1
Design Pressure	27.5 MPa (3990 psig) 110% of nominal rating
Operating Temperature	535 °F (553 K)
Inside Shell Diameter	5.322 m (209.5 in)
Shell Thickness	0.457 m (18 in)
Head Thickness	0.305 m (12 in)
Vessel Weight	1.7 million lbs

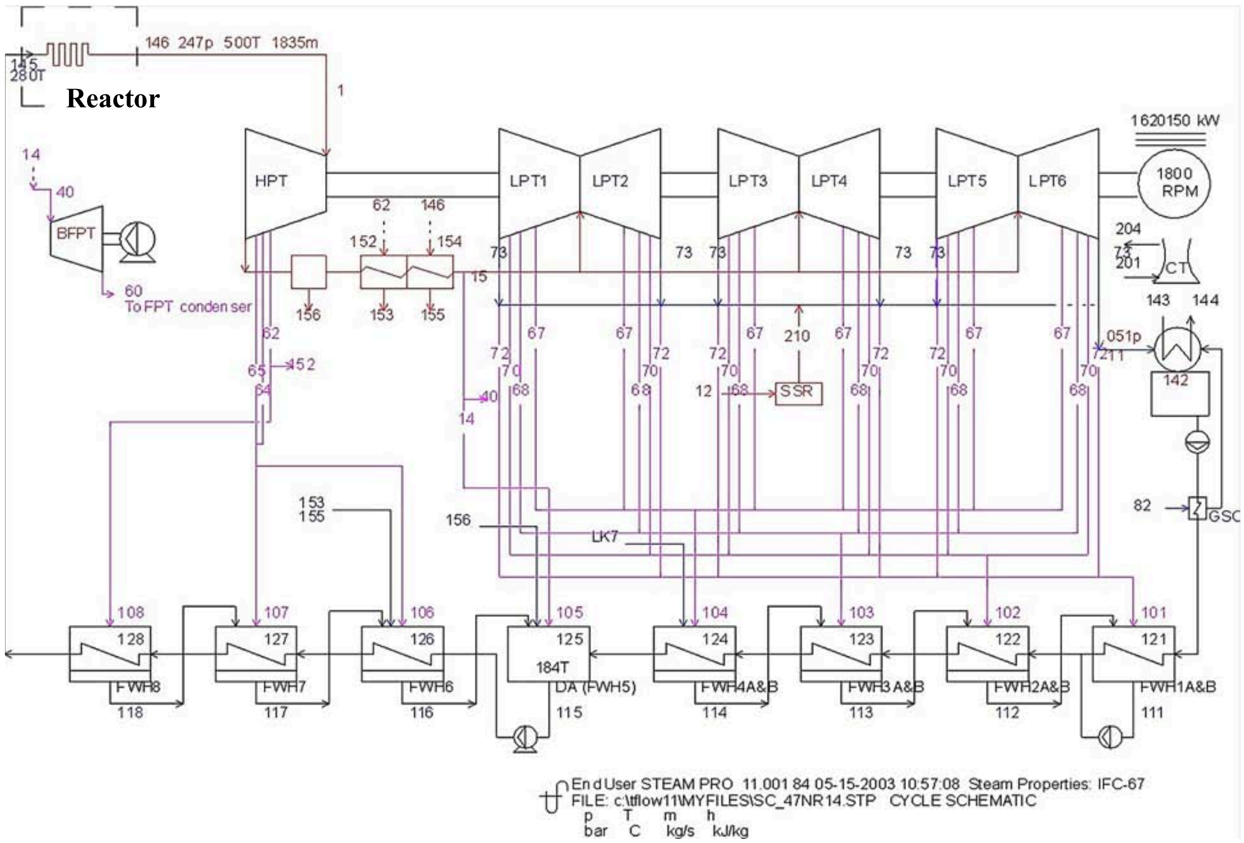


Figure 7. Schematic of the SCWR power conversion cycle (HPT = high pressure turbine, LPT = low pressure turbine, FWH = feedwater heater) [5].

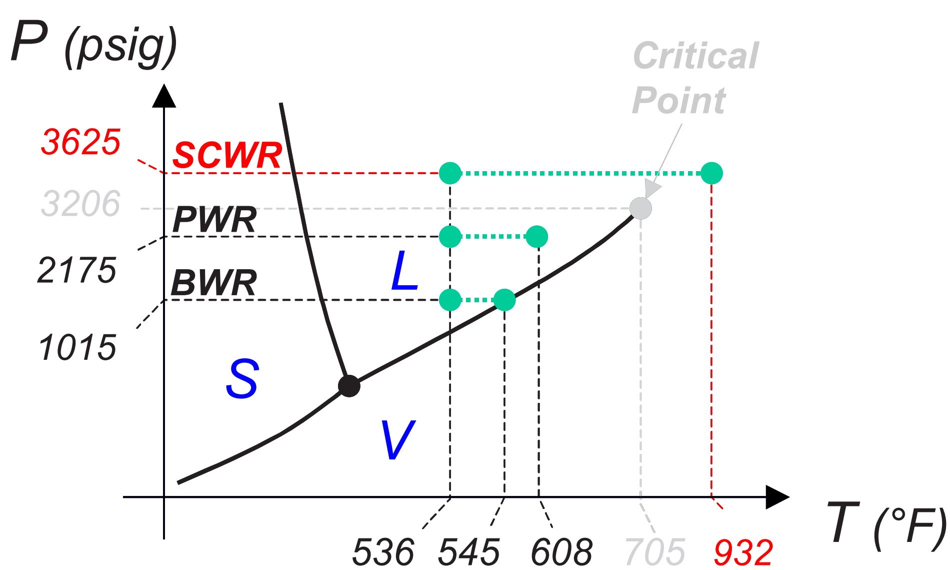
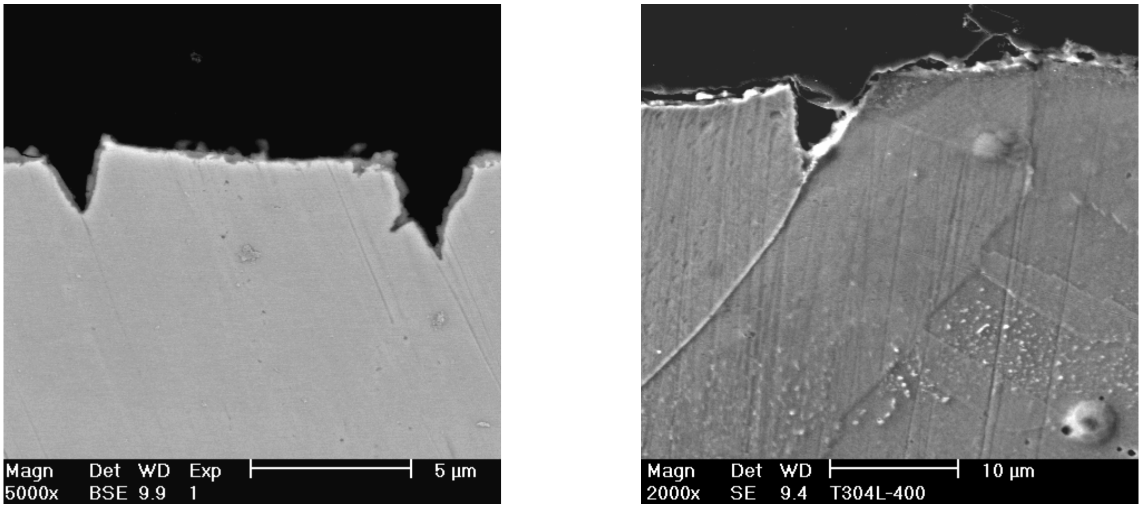
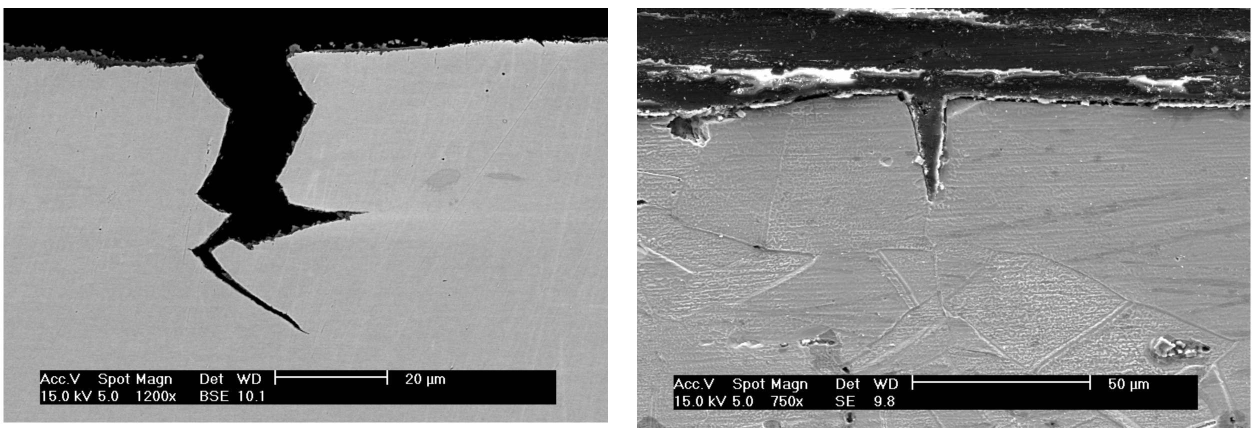


Figure 8. SCWR, PWR, & BWR operating conditions on the phase diagram of water [7].



(a)



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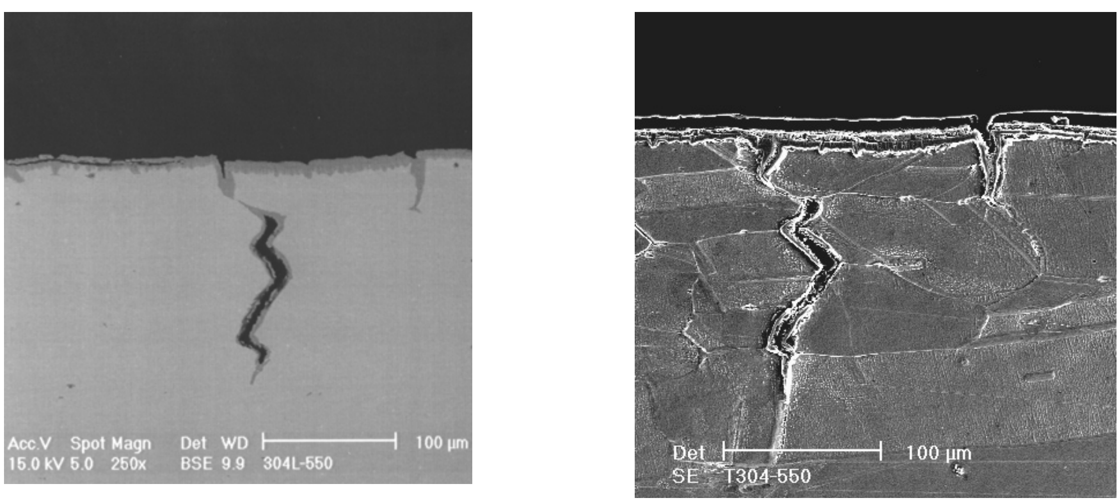

(c)

Figure 9. Cross sections of the 304L stainless steel SCC sample tested in deaerated supercritical water at (a) 750 °F, (b) 930 °F, and (c) 1020 °F [5].

& 718) based alloys which were tested in the pressure/temperature environment present in an SCWR, all were susceptible to Stress Corrosion Cracking (SCC), though Alloy 316L and Alloy 690 were sufficiently resistant to warrant further study [5]. Examples of SCC in 304L stainless steel can be seen on page 6 in Figure 9. Of the ferritic-martensitic (HT-9, T-91, & HCM12A) alloys tested, many exhibited resistance to SCC but had oxidation rates one order of magnitude more than the austenetic alloys [5]. This makes the ferritic-martensitic materials ill suited for thin assembly components. These issues provide a substantial engineering obstacle to be overcome before the SCWR can be considered a viable nuclear power plant design. Additionally, none of the material testing to date has been performed in either a high neutron or high gamma flux to characterize actual material performance in an operating reactor vessel.

Natural Gas Production with a Supercritical Geothermal Power Application By-product

In many cases, the source for Natural Gas production is a geothermally heated brine. In these cases, the brine temperatures range from 240 °F (390 K) up to 360 °F (455 K) depending linearly upon well depth [8]. In this temperature range, the brine passes through a heat exchanger to feed a supercritical Rankine cycle for Propane, which has a critical temperature of 206 °F (370 K) and a critical pressure of 616 psia. Theoretically, the brine may be able to run the cycle directly but there are too many contaminants and compositional variations for this to be feasible. If a power cycle like this were employed, the sites producing Natural Gas could potentially generate power for Grid use or, at a minimum, generate the majority of the plant's electrical requirements. In the United States, there are several areas along the Texas and the Louisiana Gulf Coast where this type of power cycle is feasible [8].

The benefit to this cycle is that it is extremely simple in terms of system components. The system requires only a single phase heat exchanger, a turbine, an air cooled condenser, and a pump. Figure 10, page 8, depicts a basic system diagram. The nominal operating pressure of this system is approximately 1000 psia, which suggests that all of the support piping and equipment is commercially available. Given a 15 Million BTU/hr brine source, this system could generate approximately 400 kW net power with a thermal efficiency of 9% [8]. Additionally, the system can be built to be self regulating by using the power grid as a dynamic brake for the turbine-generator set [8]. In effect, this acts as a speed control for the turbine during slight variations in system demand under normal operating conditions. Additional controls can be implemented to automate the system based on brine temperature and flow rates, all of which minimize the need to have a full-time operator, thus reducing operational costs.

Despite the downside of a poor thermal efficiency of only 9%, this system is still a viable source of energy. Currently, production plants are dumping the available heat to the atmosphere. The question becomes one of economics; at a cost of approximately $2,131/KW [8] (1982 dollars ≈ $5,488 today adjusted for inflation), will a 9% return pay itself back over the life of the well. At today's prices with an average electricity cost of $0.11 per kWhr, it would take approximately 6 years to recover the cost of the power plant. Since a power plant of this size can be built on a mobile platform, even if an individual well does not last 6 years, the power plant can be moved to a new location and reused. Based on the platform being reusable, this is a potentially viable power plant design despite the low thermal efficiency of 9%.

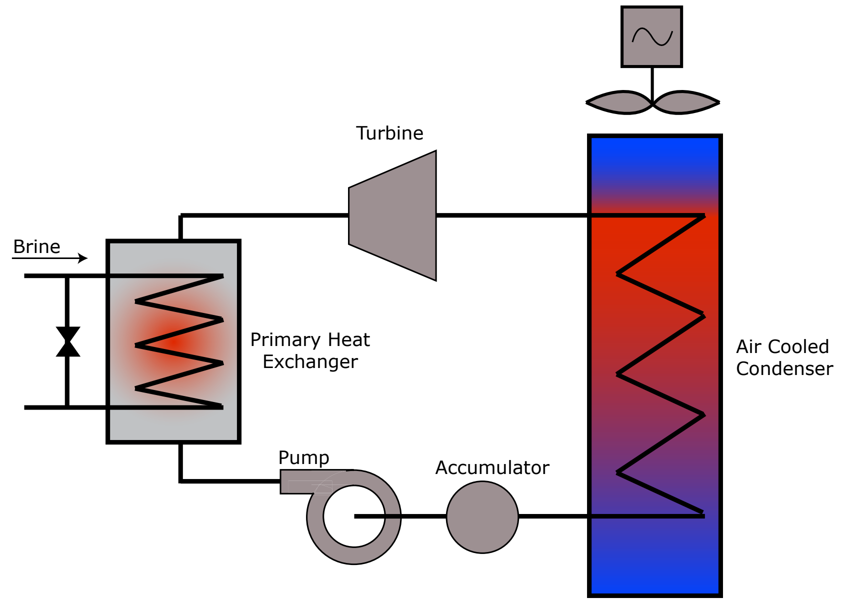

Figure 10. Supercritical Propane Cycle Fed by Geothermally Heated Natural Gas Brine.

Supercritical CO_2 Power Application

The supercritical CO_2 cycle is one which is arguably a Brayton cycle rather than a Rankine cycle. CO_2 has a critical pressure of approximately 1057 psia and a critical temperature of approximately 88 °F (305 K). Because the critical temperature is achievable during normal operating conditions, it is feasible to run along the gas/vapor boundary between the turbine outlet and the compressor inlet; thus allowing the system to maximize the work that can be extracted from the coolant. For this reason, the supercritical CO_2 cycle will be considered in this paper. A basic system diagram is shown in Figure 11, page 9, and corresponding system temperatures and pressures in table 2, page 9.

One of the Generation IV nuclear plant designs is a supercritical CO_2 cycle. This design operates at similar temperature and pressure as the SCWR. The outlet temperature is approximately 1020 °F (820 K) and the exit pressure is 2900 psia [9]. The thermal efficiency of this cycle is about 45% but due to the enhanced heat transfer properties of CO_2, the components can be made smaller and thereby reduce component costs by up to 18% compared to conventional BWR and PWR plants [9]. Also of interest, the density of CO_2 increases and compressibility decreases substantially as it approaches the critical point. This property both reduces the work required to be performed by the compressor and reduces the footprint of the physical component, thus improving efficiency and component cost. An example of the conceptual size difference can be seen in Figure 12, page 9.

Additionally, this system has reactor inlet temperatures of approximately 745 °F (670 K), which leads to temperature differentials of about 275 °F (410 K) [9]. While the conditions in the supercritical CO_2 cycle are not as extreme as those in the SCWR, they are sufficient to warrant extensive SCC tests, especially where CO_2 is potentially more corrosive than water at the nominal operating temperatures.

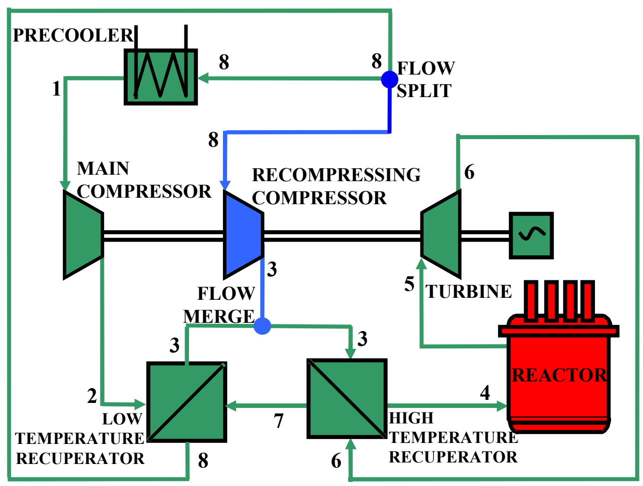

Figure 11. Supercritical CO_2 Recompression Cycle [9].

Table 2: Calculated System Parameters corresponding to Figure 11 [9].

Point	Pressure (psia)	Temperature °F
1	1101	89.6
2	2864	142.0
3	2862	316.4
4	2858	745.8
5	2839	1022.0
6	1131	824.5
7	1119	335.0
8	1103	157.3

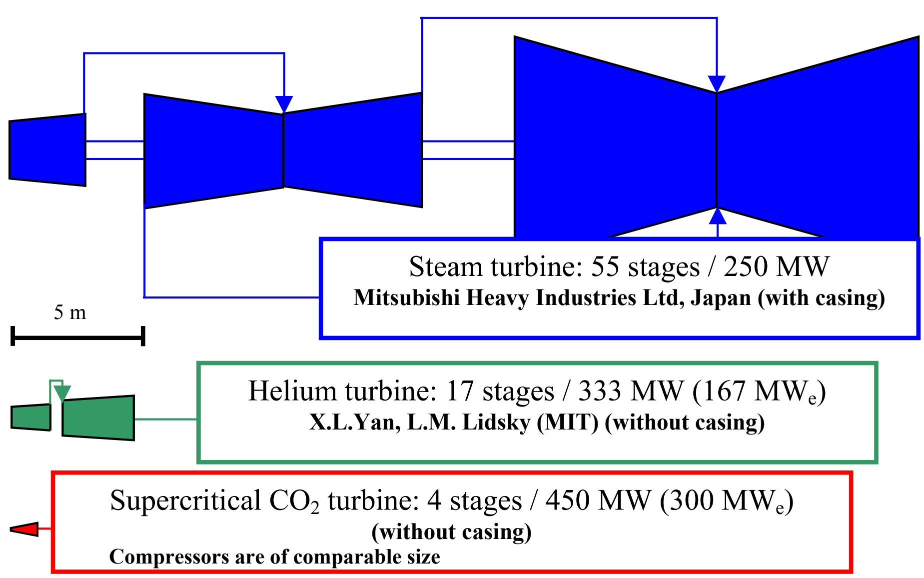

Figure 12. Size Comparison between Turbines of Various Cycles/Coolants [9].

The supercritical Rankine cycle, in general, offers an additional 30% relative improvement in the thermal efficiency as compared to the same system operating in the subcritical region. The cycle has been successfully utilized in fossil fuel plants but the current available materials prohibit reliable application of the supercritical cycle to nuclear applications. There is much work to be done in order to advance materials to the point where they will be able to reliably withstand the stresses of a supercritical environment inside a nuclear reactor for a designed life span of approximately 60 years.

While many of the material advances are due to the advent of specialized coatings, it is reasonable to suspect that new advances could also be made by improvements in the isotopic quality of the base metals. It has been known for decades that isotopically pure materials have considerably better thermal conductivity. Improvements in isotopic purity can affect heat transfer characteristics by up to a factor of three [11], possibly more. It should be noted however, the cost of obtaining sufficient quantities of such materials may be prohibitive and the benefit gained is temperature dependant.

One other consideration for a supercritical Rankine cycle is the size of the plant. For large plants on the order of a 1600 MW_e the thermal efficiencies are in the 40% region. By the time the power plants are reduced to the 20 MW_e range, thermal efficiencies are down into the low to mid 20% region [7]. As indicated by the geothermal propane application, when a plant is in the 400 kW_e region, the thermal efficiency is lower yet. For the propane cycle, it was 9%; by extrapolation, a supercritical steam cycle would be ≈ 19% efficient were it applicable. The issue for smaller systems is the cost effectiveness. A supercritical cycle is simpler and reduces the amount of equipment required to operate the cycle but because it operates at much higher pressures and temperatures the cost of the equipment which is required, goes up considerably. Small systems may have improved thermal efficiency but may not be the most cost effective solutions; for this reason they should be considered skeptically before being implemented. Just because a system can be made more efficient does not mean it is the best allocation of money, or material resources.

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